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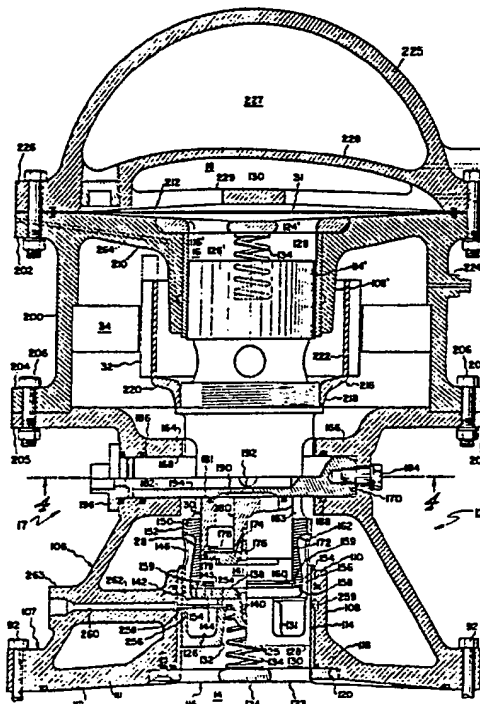
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(54) Title: HERMETIC RESONANT PISTON STIRLING ENGINE COMPRESSOR ALTERNATOR HAVING HYDRAULIC COUPLING DIAPHRAGM

(57) Abstract

A free-piston Stirling engine usable as a heat pump has a closed vessel filled with helium working gas which is heated at the bottom end and cooled at the top end. The vessel contains a displacer (22) supported for axial reciprocal oscillation on a gas spring post (78) mounted on the vessel. The displacer (22) shuttles the working gas from end to end in the vessel, alternately heating and cooling the gas. The vessel is sealed with a flexible diaphragm (26) which flexes in response to the pressure wave generated in the vessel as the working gas is alternately heated and cooled. When the diaphragm (26) flexes, it displaces hydraulic fluid in a hydraulic chamber (14) and drives a power piston (126) for driving a linear alternator and a gas compressor. A gas spring (18) operating on a second hydraulic cylinder (128) on the other side of the power piston (126) stores part of the energy of the piston stroke and returns it for the return stroke. Controls (232, 290) are provided for balancing and controlling the hydraulic fluid pressure, for starting the Stirling engine, and for modulating its power output.



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1 HERMETIC RESONANT PISTON
 STIRLING ENGINE COMPRESSOR ALTERNATOR HAVING
 HYDRAULIC COUPLING DIAPHRAGM

TECHNICAL FIELD

5 This invention relates to an improved Stirling
 engine and in particular to an improved free-piston
 Stirling engine having a hydraulic coupling to an
 output member such as a compressor of a heat pump.
 This application is related to U.S. Serial No.
10 168,714 filed 7/14/80 for "Heat Engine Device," filed
 by Harlan V. White, filed concurrently herewith the
 disclosure of which is incorporated herein by refer-
 ence.

BACKGROUND ART

15 The Stirling engine is a closed-cycle engine
 with cyclic recirculation of the working fluid.
 Power is produced by compressing the working fluid
 at a low temperature and expanding it at a high tem-
 perature. The required heat is added continually
20 during expansion of the working gas inside the engine
 through a heat exchanger wall. Since this wall has
 a high heat capacity, it is not possible to rapidly
 heat and cool the same wall; therefore, the working
 gas is alternately shuttled between two stationary
25 variable volume chambers in the working space, held
 respectively at high and low temperatures and called
 the hot space and the cold space.

 The alternating heating and cooling of the same
 working gas would inherently waste quantities of
30 heat, so a regenerator is placed between the hot and
 cold sources in the path of the working gas. Heat



-2-

1 is stored in the regenerator as the gas moves from
the hot space toward the cold space and is then re-
leased to the working gas as it passes back through
the regenerator in moving from the cold space to the
5 hot space.

The conventional Stirling engine includes two
pistons: one, called the displacer, is a lightweight
body mounted on a rod which moves the displacer to
shuttle the working gas between the hot and cold
10 spaces; the other, called the power piston, is of
heavier construction and is responsible for the work
transfer over the cycle.

The motions of the power and displacer pistons
can be considered from a first order perspective, to
15 give rise to three pressure wave components, two of
which occur within the cold space or engine compres-
sion space. The first pressure component, called the
power piston pressure wave, is associated with the
motion of the power piston. Physically, this is the
20 pressure wave which would exist in the engine if the
displacer piston were held fixed and the power piston
were oscillated sinusoidally. The amplitude of the
power piston pressure wave is related to the springi-
ness of the engine and is primarily a function of the
25 engine pressure, enclosed volume, piston area, and
piston mass.

The second component is associated with the mo-
tion of the displacer piston and is called the dis-
placer piston pressure wave. Physically, this is the
30 pressure wave which would exist in the compression
space volume if the power piston were fixed and the
displacer piston were oscillated sinusoidally. This
wave is the result of two generally conflicting



-3-

1 effects: the first is the change in pressure which
results from moving the displacer rod in and out of
the engine volume; the second is the change in pres-
5 sure which occurs due to the change in gas tempera-
ture as the working fluid is shuttled between the
hot and cold spaces in the engine. As the displacer
moves toward the engine hot space, the first effect
tends to increase the pressure and the second effect
tends to decrease the pressure. For any practical
10 engine operating point, the temperature effect more
than offsets the volume effect. As a consequence,
the displacer pressure wave leads the displacer mo-
tion by 180°.

The third component of the pressure wave occurs
15 in the expansion space volume or the hot space and is
due to seal leakage. This component results from the
pressure drop across the seal and is proportional to
the pressure amplitude, leading the pressure by 90°.
It is inimical to good engine efficiency and is the
20 subject of considerable development effort to mini-
mize. The sum of these three components is the pres-
sure wave in the working space; if there were no
pressure drop in the heat exchanger duct, this wave
would lead the power piston motion.

25 The pressure wave components in the expansion
and compression spaces may further be broken down in-
to two elements: first, the basic pressure wave and,
second, the pressure drop due to flow through the heat
exchanger duct. The basic pressure wave approximates
30 the pressure wave which would be measured in the mid-
dle of the heat exchanger duct. The expansion space
pressure is the basic pressure plus the pressure drop
between the middle of the heat exchanger duct and the
expansion space. The compression space pressure is



1 found in a similar manner. The forces which are
exerted on the power piston and the displacer, because
of the basic pressure wave, are obtained by multiply-
5 ing the magnitude of this pressure wave by the area of
the power piston face and the displacer rod area, re-
spectively. These forces, which are 180° out-of-
phase with the pressure wave, are in a ratio of ap-
proximately 10:1. The power is proportional to the
10 component of the force phasor which is normal to the
displacement vector. As a consequence of the dis-
placer rod area, the engine does feed power into the
displacer through the rod area, and if the rod area
is large enough, this power will exceed the power
dissipated through the heat exchangers. The lag angle
15 between the engine pressure wave and the power piston
phasor is referred to as the engine pressure angle.
A low-pressure angle corresponds to a peaked or
springy PV diagram while a high-pressure angle cor-
responds to a more oval or flat PV diagram. From a
20 thermodynamic perspective, a flat PV diagram is more
desirable than a peaked PV diagram since the flat
diagram has a lower peak-to-peak pressure ratio and,
hence, a smaller temperature variation of the gas in
the compression and expansion space volume, and there-
25 fore, lower thermal mixing and thermal energy losses.
The thermal mixing loss is the irreversibility which
occurs when gas from the heater or cooler enters the
expansion or compression space volume at a tempera-
ture significantly different from the gas temperature
30 within the volume. The thermal entry loss is the
loss which occurs when gas from the expansion or com-
pression space enters the heater or cooler at a tem-
perature significantly different from the heater or
cooler metal temperature.

1 The unique feature of free-piston Stirling engines is that the piston motions are determined by the state of a balanced dynamic system of springs and masses, rather than a mechanical system.

5 The free-piston Stirling engine is an ideal vehicle to power residential-sized heat exchangers. It is extremely quiet, indeed virtually silent, in operation. It can be designed to be heated by any fuel whatsoever, and therefore is able to utilize the
10 cheapest and most available fuel at any particular time. By using the same fuel for both heating and cooling, the seasonal demand on particular power sources can be substantially leveled to the benefit of the distribution system. The free-piston Stirling
15 engine is sealed so the working fluids within the pressure vessel are not subject to loss through the shaft seals of conventional mechanical output Stirling engines. However, in a closed hermetic system utilizing more than one working fluid, it is necessary
20 that they be separated. In addition, the lubrication within the sealed vessel must be maintained at the correct pressure and properly separated from the other working fluids, particularly the engine working fluid.

25 Power modulation of a Stirling engine heat pump alternator is theoretically controllable by controlling the pressure of the working gas in the engine. However, this also has the effect of altering the engine frequency which in turn can alter the frequency of the electric output of the system. In some situations,
30 it may be desirable to regulate the power output while maintaining the system frequency constant.

1 As the power requirements for the heat pump in-
crease in hot or cold weather, this condition must be
sensed by the system which must automatically adjust
the operating parameters to produce a higher output
5 power. The conventional technique for accomplishing
the power modulation is to adjust the time interval
in which the compressor operates. This is inherently
inefficient because of start-up power surges and the
other known inefficiencies in operating a high-power
10 output device intermittently to produce low power out-
put levels. A much more efficient method would be to
run a system continuously but modulate the input
energy to produce a controlled output power as
desired.

15 Disclosure of the Invention

It is an object of this invention to provide a
free-piston Stirling engine having a displacer sprung
to ground and a hermetic separation of the engine
working fluid and the power piston. The engine is
20 coupled to a gas compressor having a P-V diagram ro-
tated 90° to the engine P-V diagram, so an energy stor-
age mass is incorporated in the compressor to absorb
energy from the engine during the high power output
portion of its cycle, and to deliver that stored
25 energy to the compressor during the high power demand
portion of its cycle.

A power diaphragm is provided to seal the work-
ing gas in the working space, and a hydraulic coupling
between the power diaphragm and the power piston pro-
vides a uniform backing for the diaphragm and a means
30 for selecting the stroke of the power piston. The
displacer is supported for axial oscillation on a
post fixed relative to the vessel and incorporates a

-7-

- 1 gas spring between the post and the displacer. A mid-
stroke porting arrangement is provided to maintain the
equality of the gas spring and working gas mean pres-
sures. A control is provided for continuously modu-
5 lating the engine power in response to system demands.

Brief Description of the Drawings

- The invention and its many objects and advan-
tages will be come more clear upon reading the follow-
ing description of the preferred embodiment in con-
10 junction with the following drawings, wherein:

Figs. 1A and 1B are the top and bottom sections,
respectively, of a sectional elevation of a Stirling
engine driven alternator/compressor made in accordance
with this invention;

- 15 Fig. 2 is a plan view along lines 2-2 in Fig. 1B;

Fig. 3 is an exploded view of portions of the
gas compressor piston cylinder assembly shown in
Fig. 1A;

- 20 Fig. 4 is a sectional plan view along lines 4-4
in Fig. 1A;

Fig. 5A is a combined temperature-entropy graph
of the engine and the compressor of the embodiment
shown in Fig. 1;

- 25 Fig 5B is a combined pressure volume diagram of
the engine working gas and the compression spaces in
the compressor;

Fig. 5C is a schematic diagram of the Stirling
engine and compressor of the embodiment shown in
Fig. 1; and

- 30 Fig. 6 is a schematic view of the controls for
the device shown in Fig. 1.



1 Description of the Preferred Embodiment

Referring now to the drawings wherein like reference characters designate identical or corresponding parts, and more particularly to Fig. 1 thereof, a
5 Stirling engine powered alternator-compressor is shown having a Stirling engine working section 10 and a power section which includes a compressor-alternator assembly 12. The power section and the working section are coupled through a lower hydraulic chamber 14.
10 The distal end (the top end in Fig. 1A) of the compressor-alternator assembly 12 is coupled through an upper hydraulic chamber 16 to a bounce space 18. The working section 10 and the alternator-compressor are all enclosed within a hermetically sealed vessel
15 17 having a vertical axis.

Broadly, the energy flow through the system begins with heat input to the heater head 20 of the engine which heats a charge of working gas contained within the working space of the engine working section 10. A displacer 22 moves axially in a reciprocating manner in the working space and causes the
20 working gas to cycle between the hot end defined by the heater head 20 and the opposite end which is kept cold by a cooler 24. The cyclic heating and cooling
25 of the working gas causes a periodic pressure wave which is transmitted through a flexible engine diaphragm 26 to the hydraulic fluid in the hydraulic chamber 14 where it drives a compressor cylinder 28 to compress a gas or vapor such as Freon refrigerant
30 in conjunction with a fixed piston 30. The other end of the compressor cylinder 28, operating in a hydraulic chamber 16, is similar in shape to the first mentioned end of the compressor cylinder operating in the hydraulic chamber 14 and is coupled through

1 the hydraulic fluid in the chamber 16 and a bounce
diaphragm 31 to the gas spring bounce space 18. An al-
ternator armature 32 is fastened to the compressor
cylinder 28 and oscillates with it opposite a fixed
5 alternator stator 34 to produce electrical output
power.

The engine displacer 22 oscillates axially in a
working space which is defined by the inner surfaces
of the heater head 20, a cylindrical regenerator
10 housing 36 to which the heater head 20 is screwed, a
cylindrical cooler housing 38 to which the regenera-
tor housing is attached by bolts 40, a base member 42,
and the engine diaphragm 26. A shell 46 is anchored
to the base 42 at 47 and extends downwardly therefrom
15 coaxially through the working space. The shell 46
has an axial opening 48 in the lower end thereof for
directing the flow of working gas in close proximity
to the heater head for the purpose of heating the gas.

The displacer 22 includes a top portion 50 having
20 an outside cylindrical wall 52, a flat radially ex-
tending annular roof 54 and coaxial sleeve 56 extend-
ing into the middle of the displacer 22. The end of
the sleeve 56 is enlarged and closed by an end wall 57
to form a chamber 58. The bottom end of the cylin-
25 drical wall 52 terminates in an inwardly extending
radial flange 60 and a small axially extending lip 62
to which is fastened the top edge of a cylindrical
body 64 having a closed rounded bottom end 66. Three
disc-shaped stiffener elements 68 to hold the shape
30 of the cylindrical body 64 and act as heat shields.
The center of each of the stiffener members 68 is
punctured by a small hole 70 for enabling the interior
of the displacer to pressurize equally throughout.



1 The base 42 includes a plinth 72 fastened by an
axial screw 74 to the central portion of the base 42
and held against rotation with respect thereto by a
locating pin 76. An integral axial post 78 depends
5 from the plinth 72 in snug fitting relation to the
sleeve 56. The interior of the axial post 78 is pres-
surized, by a system which will be described below,
and provided with a series of small holes 79 which ad-
mit pressurized gas to the interface between the post
10 78 and the sleeve 56 to act as a gas bearing. A hole
80 is drilled in the sleeve 45 axially midway between
the series of holes 79 to act as a gas bearing drain.

 The regenerator housing 36 encloses an annular
cylindrical regenerator 81 composed of a network of
15 fine high-temperature wires such as Nichrome or
Inconel. The wires are arranged in a screen or mesh
configuration which presents a minimal impedance to
the gas flow through the regenerator while presenting
a substantial surface area to the gas to facilitate
20 the heat exchange between the wire and the gas. The
connection between the regenerator housing and the
heater head 20 and the cooler 24 facilitates easy in-
spection and replacement of the regenerator should
that be necessary.

25 The cooler 24 includes the cooler housing 38 pre-
senting an annular space between the inner surface of
the cooler housing 38 and the outer surface of the
shell 46. A cooler assembly is secured in the annu-
lar space and sealed therein as by brazing or other
30 secure means to prevent any mixing between the coolant
and the engine working fluid. The cooler is an annu-
lar assembly having a top plate 84 and a bottom plate
85, both of which are perforated with a multitude of



-11-

1 closely spaced small diameter aligned holes. A
plurality of fine tubes 86 are brazed at their ends
between the top and bottom plates 84 and 85 to pro-
vide a gas passage between the plates with a very
5 large heat exchange surface area. A set of three
radial baffles 87 is arranged between the top and
bottom plates and alternately fastened at outer and
inner circumferential edges to the cooler housing 38
and shell 46, respectively, thereby providing a ser-
pentine path for the coolant. An inlet connection 89
10 and an outlet connection 89' are provided for con-
nection to coolant lines for circulation of a liquid
coolant such as water or liquid Freon through the
cooler and an external heat exchanger (not shown) for
15 effective cooling of the working gas. The baffle
arrangement makes maximal use of the coolant by
creating high rates of flow around the tubes and a
multi-pass, counter-current flow for optimal heat
exchange.

20 The base member 42 includes an outer flange 88
having holes 90 formed therethrough for receiving
bolts 92 which fasten the base 42 to the power sec-
tion 12 of the vessel. The top face of the base 42
is formed in a shallow concavity 94 which, with the
25 engine diaphragm 26, forms a portion 95 of the cold
space or engine working gas compression space. The
face 94 also serves as a limit surface to prevent ex-
cessive downward deflection of the diaphragm 26.

The base member 42 includes a series of passages
30 96 extending completely through and establishing com-
munication between the cooler and the portion 95 of
the compression space. A second set of passages 98
formed through the base 42 at a position radially



-12-

1 inward of the passages 96 establishes communication
between the portion 95 of the working gas compression
space and a second or lower portion 100 of the working
gas compression space bounded between the top face
5 102 of the displacer and the bottom face 104 of the
plinth 72.

The diaphragm 26 lies in a plane which is per-
pendicular to the axis of vessel 17 and approximately
coaxial therewith. This has the advantage of great
10 compactness and rugged construction. It is made pos-
sible because the displacer drive arrangement is lo-
cated within the displacer and therefore does not
require external driving mechanisms. An additional
advantage of this arrangement, as will appear more
15 clearly in the following description, is the compact
connection of the power section, directly to the same
vessel with the engine. This makes possible a low
cost, unitary power module in which all power trans-
mission is within the vessel and the only connections
20 to the vessel are fuel lines and power take-off lines
from the compressor and alternator.

The gas flow path of the Stirling engine will
now be described in connection with the theoretical
or ideal Stirling engine cycle as shown in the
25 temperature-entropy and pressure-volume diagrams of
Figs. 5A and 5B. At an arbitrarily selected starting
point, the displacer 22 is at its lowermost position
with the dome-shaped bottom portion of the displacer
close to the dome-shaped bottom portion of the shell
30 46, and the engine diaphragm 26 in its uppermost po-
sition away from the top face 94 of the base member 42.
In this configuration, the gas volume is maximum and
the gas temperature and pressure are minimum. This

-13-

1 is point A on the pressure-volume and temperature-
entropy diagrams. The process A-B is performed by
the diaphragm 26 moving downwardly toward the face 94
of the base member 42. This process is an isothermal
5 compression of the cold working gas wherein the heat
occasioned by the compression of the gas is trans-
ferred from the gas to the cooler 24. At position B
the diaphragm is at its lowermost position against
the face 94 of the base 42 and the displacer is at its
10 lowermost position with the rounded end 66 of the dis-
placer 22 close to the rounded bottom end of the shell
46. At this point, the volume, temperature, and en-
tropy are all at their minimum values.

The next motion is the motion of the displacer
15 moving upwardly away from the curved bottom face of
the shell 46 toward the bottom face 104 of the plinth
72. This displacer motion causes the gas to be dis-
placed from the cold space at the top end of the en-
gine through the cooler and then through the regenera-
20 tor where it is reheated by the heat deposited in the
regenerator during the last cycle, and then passes
between the heater head 20 and the shell 46 where it
is heated at constant volume by heat transfer from the
heater head 20. When the displacer 22 reaches its
25 uppermost position, the gas, still at minimum volume
but now at maximum pressure and temperature, does work
on the diaphragm, driving the diaphragm upward in the
process C-D which is an isothermal expansion of the
working gas where heat is transferred to the working
30 fluid at the maximum cycle temperature of the external
source. The displacer again moves downward to dis-
place the hot gas toward the cold zone, during which
it passes through the regenerator and deposits its
heat into the regenerator where it is cooled at



1 constant volume. This is the process D-A in which the
pressure and temperature drop at constant volume. The
cycle then repeats itself at the natural frequency of
the system. If the heat which is transferred to the
5 working fluid from the regenerator matrix is the same
as that transferred from the fluid to the matrix, then
only the external heat transfer processes remain; the
efficiency is consistent with the Carnot cycle effi-
ciency. The advantage of the Stirling engine cycle
10 is that the two isentropic processes of the Carnot
cycle are replaced by two constant volume processes
which increase the area under the P-V diagram result-
ing in higher specific work output levels without re-
sorting to very high pressures and high swept volumes.

15 The ideal cycle described assumes that the proc-
esses are thermodynamically reversible. That is, the
expansion and compression processes are isothermal
and that infinite heat rates exist in addition to in-
finite heat capacities. The ideal analysis neglects
20 the effects of regenerator matrix voids, clearance
spaces and cylinder pockets. In addition, the dis-
placer and engine diaphragm are assumed to move in a
discontinuous manner whereas, in reality, the motion
is smooth and continuous. Therefore, the theoretical
25 P-V and T-S diagrams are rounded off as shown in the
oval-shaped curves. Aerodynamic and mechanical losses
are also neglected. Inclusion of these losses, of
course, results in a lower net cycle output power and
lower efficiency. The addition of heater and cooler
30 components changes the real heat transfer to a more
adiabatic situation rather than the assumed isothermal
processes. Penalties in additional aerodynamic flow
losses and increased dead volume result. The use of
practical equipment imposes one additional reality;
35 that the fluid is heated not only as it flows to the

1 expansion space, but also as it flows in the reverse
direction from the expansion space to the cooler.
The cooling process is also penalized in this manner
as well. These losses have been minimized by this
5 engine design to maximize the engine efficiency toward
the ideal Carnot efficiency.

Turning back again to Fig. 1, the power section
will now be described. The hermetic vessel 17 includes
a cast aluminum casing 106 having a lower flange
10 107 to which the outer flange 88 of the base member
42 is bolted by the bolts 92. The casing 106 includes
an integral hydraulic cylinder 108 which is coaxial
with the vessel axis. The hydraulic cylinder 108
terminates at a top end 110 and flares on its bottom
15 end in a web 111 with a concave face extending outward
to the same diameter as the concave face 94 on the
base member 42. The bottom face 112 of the casing 106
serves a function corresponding to the concave face 94
of the base 42, namely to prevent excessive deflec-
20 tion of the engine diaphragm 26 and also to provide a
wide area over which the engine working gas can act
through the engine diaphragm 26 to displace a sig-
nificant volume of hydraulic fluid to act in the hy-
draulic cylinder 108.

25 The hydraulic cylinder 108, which is cast in-
tegrally with the case 106 and therefore is of the
same material, is lined with a sleeve 114 of high-
strength, wear-resistant material such as stainless
steel. The sleeve is retained in position by a spi-
30 der 116 having an outer ring 118 which fits into a
recess 120 at the lower end of the hydraulic cylin-
der 108. The spider also includes a series of arms
122 extending inwardly to a center disc 124.

-16-

1 A piston assembly 125, shown exploded in Fig. 3,
is mounted for axial reciprocation in the vessel 17
and includes a cup-shaped lower end member 126 at-
5 tached to the cylinder 28 and operating in the hy-
draulic cylinder 108. The piston lower end member
126 includes a lower working face 128 which, with the
top face of the engine diaphragm 26, defines the lower
hydraulic chamber 14. The flexing of the diaphragm
26 in response to a pressure wave generated in the
10 working gas in the Stirling engine working space dis-
places hydraulic fluid upwardly in the hydraulic cham-
ber 130 to drive the piston 126 upwardly in the hy-
draulic cylinder 108.

 The piston assembly 125 is substantially symmet-
15 rical about the transverse plane 4-4, except for an
elongation of the top piston member 126' to provide a
threaded mounting collar for the linear alternator
armature, as will be described below. A hydraulic
cylinder 108' lined with a sleeve 114' is disposed at
20 the top end of the vessel 17 to receive a cup-shaped
top end member 126' of the piston 125. The end face
128' of the top end member 126' of the piston 125,
along with the bounce diaphragm 31, defines a top hy-
draulic chamber 130' which cooperates with a bounce
25 space 18 to balance the piston 125 dynamically so
that it will produce two power strokes for each cycle
of the engine working space. The following detailed
description of the piston assembly 125 lower end will
be understood to apply also to the symmetrically iden-
30 tical structure of the top end, therefore the descrip-
tion will not be repeated for the top end.

 An axial boss 131 is formed inside the piston
end member 126 extending inward, away from the piston



-17-

1 face 128. The boss is hollow, defining an axial well
132 opening in the face 128 of the piston. The well
132 receives one end of a centering spring 134 which
is biased between the end wall of the well 132 and
5 the top face of the disc 124. A similar centering
spring 134' acts in a symmetrically identical well
opening in the top end 126' of the piston 125 to exert
a centering force on the piston in its cylinders 108
and 108'.

10 An inlet valve seat disc 138 is fastened to the
top end of the central boss 131 by a screw fastener
140 or the like. The diameter of the inlet valve
seal disc 138 is smaller than the inner diameter of
the piston lower end member 126 to provide an annular
15 passage 142 through which gas to be compressed can
flow into the compression space, as will be described
below. A series of inlet openings 144, formed in the
inlet valve disc, are controlled by an annular valve
reed 145 for admitting gas to be compressed and pre-
20 venting the exodus of gas from the compression cham-
ber as it is compressed, all to be described presently.
A series of apertures 146 is formed through the side-
wall of the piston lower end member 126 for the pur-
pose of admitting gas to be compressed into the com-
25 pression chamber.

The top inside periphery of the piston lower end
member 126 is internally threaded at 150 and screwed
onto a threaded portion 152 of the cylinder 28. The
cylinder 28 is threaded to and becomes part of the
30 piston 125, but it functions as a moving cylinder re-
ciprocating on the fixed piston 30. Thus the piston
125-cylinder 28 assembly has both piston and cylinder
functions. The cylinder 28 includes a lower end 154



-18-

1 having a diameter identical to the diameter of the
inlet valve seat disc 138. An annular groove 156 in
the end face of the cylinder 28 receives an O-ring
158 to seal the junction of the cylinder 28 to the in-
5 let valve seat disc 138. The inside lower edge of the
cylinder 28 is notched at 159 to receive a stop ring
160 which extends over the inlet valve reed 145 to
retain the reed during the suction stroke in which gas
flows through the inlet openings, around the reed 145,
10 and into the compression chamber 161. The end of the
cylinder 28 is necked down at 162 in the vicinity of
the apertures 146 in sidewall of the piston end mem-
ber 126 to provide the aforementioned annular gas pas-
sage 142 for gas passing from the interior of the cas-
15 ing 106 into the compression space. An oval slot 163
extends through both sides of the central section of
the cylinder 28 between the threaded portions 152 and
152' to provide clearance for the cylinder 28 to re-
ciprocate around the pipe 182, to be described below.

20 A cylindrical boss 164, best shown in Fig. 4, is
formed in the side of the casing 106 at the center of
the midstroke position of the piston 125. A corres-
ponding boss 166 is formed in the wall of the casing
106 diametrically opposite the boss 164. A bore 168
25 extends completely through the boss 164 perpendicular
to the vessel axis, and an aligned bore 170 extends
partially into the boss 166.

The fixed piston 30 is disposed in the center of
the cylinder 28. The fixed piston 30 includes a pis-
30 ton body 172 and an exhaust valve disc 174 screwed to
the end of the piston body 172 by a screw 176 or the
like. An annular exhaust valve reed 178 lies over
a series of openings 179 in the exhaust valve



-19-

1 disc 174 for exhausting gas which has been compressed
between the inside face of the inlet valve disc 138
and the outside face of the exhaust valve disc 174,
which space constitutes the compression space 161 of
5 the gas compressor. In a manner similar to the inlet
valve arrangement, the lower end of the piston body
172 is notched to receive a valve stop ring which is
held in place in its notch by the outlet valve disc
174. A series of gas channels 180 communicate between
10 a plenum behind the exhaust valve reed 178 and a cen-
tral transverse bore 181 extending to the bore 168 in
the boss 164 and the bore 170 in the boss 166.

A pipe 182 extends through the bore 168 in the
boss 164, the slots 163 in the cylinder 28, the bore
15 181 in the piston body 172 and into the bore 170 in
the boss 166. A screw 184 is threaded into an in-
ternally threaded hole in the end of the pipe 182 to
fasten the pipe to the casing 106. An O-ring 186
seals the pipe 182 in the bore 168 of the boss 164
20 and a corresponding O-ring 188 seals the pipe in the
bore 170 of the boss 166. The pipe 182 extends
through the central transverse bore 181 in the piston
body 172 to secure the piston 30 in place in the cas-
ing 106 and to establish fluid communication from the
25 interior of the piston 30 to the exterior of the case
106. This communication is established by a recess
190 which connects the channels 180 in the piston
body 172 with an aperture 192 in the pipe 182, which
aperture permits gas to flow from the piston interior
30 into the interior 194 of the pipe 182. A fitting 196
is threaded into an internally threaded portion at
the end of the boss 164 for connection to external
gas lines by which the compressed gas may be piped
to its use, as in a heat pump.

1 The manner of assembly of the compressor apparatus will now be described. The exhaust valve discs 174 and 174' are screwed to the ends of the piston body 172 which is then inserted into the cylinder 28.

5 The inlet valve seat discs 138 and 138' are screwed to the interior top of the central boss 131 on the two piston end portions 126 and 126', respectively, and the two piston end portions 126 are screwed onto the cylinder 28 at the threaded portion 152. The

10 lower end 126 of the piston cylinder assembly is then inserted into the hydraulic cylinder 108 and the pipe 182 is slid through the bore 168, the slots 163 in the central portion of the cylinder 28, the bore 181 in the piston body 172, and into the bore 170. The screw

15 184 is threaded into the threaded hole in the end of the pipe 182 to secure the assembly into position.

 The diaphragms 26 and 31 have a number of functions in the system. One important function is providing a hermetic and thermal separation of the engine

20 and compressor working gases. Any intermixing of the working gases would have a deleterious effect on the performance of the engine or the compressor because of the particular characteristics of the working gases in the thermodynamic cycles they perform. It is desirable, therefore, that there be a "hard" or hermetic

25 separation of the working gases, and this precludes the use of sliding seals. One technique for providing the hermetic separation of working gases in an internal compressor is the spring tube, disclosed in the

30 aforementioned U.S. Application Serial No. 168.7.4. While this arrangement works well, there is a thermal penalty introduced by the close proximity of the engine working gas to the compressor working gas across a metal interface of large surface area. The engine



1 working gas in the cold compression space is consider-
ably hotter than the compressor working gas at suc-
tion pressure, and the transfer of heat through the
spring tube to the compressor suction gas imposes an
5 efficiency penalty to the compressor. The design of
the invention disclosed herein substantially reduces
that heat transfer and thereby improves the effi-
ciency potential of the compressor.

10 The hydraulic coupling between the diaphragms
and the compressor provides an ideal backing for the
diaphragms by eliminating stress concentrations and
also provides an ideal environment in which the pistons
126 and 126' can operate with low friction and uni-
form temperature. This hydraulic fluid would cause
15 severe problems if it leaked into the engine, but
such leakage is positively prevented by the hermetic
sealing of the diaphragms.

20 This design has two degrees of freedom which is
a simple arrangement to control, thereby simplifying
the system controls. The controls are described be-
low and will be seen to be uncomplicated, inexpensive
and reliable fluid controls. This simplification is
made possible by the use of diaphragms which eliminate
at least one degree of freedom in the system.

25 An alternator housing 200 having top and bottom
flanges 202 and 204 is secured to the casing 106 by
means of bolts 206 which secure the bottom flange 204
to a top flange 205 of the case 106. The housing 200
includes a depending hydraulic cylinder 108' con-
30 nected to a top web 210 of the housing 200. The top
surface 212 of the web 210 is slightly concave to pro-
vide a backstop for the bounce space diaphragm 31.



1 An upper end hydraulic chamber 130' is defined between
the diaphragm 31, the top surface 212 of the web 210,
the interior of the hydraulic cylinder 108', and the
top face of the upper piston end member 126'. The
5 function of the hydraulic chamber 130' in conjunction
with the bounce space 18 will be described below.
Except for the linear alternator armature mounting
ring, the top end portion of the piston cylinder as-
sembly is symmetrically identical to the lower end
10 portion.

A linear alternator is mounted in the housing
200 for generating electrical power. The alternator
includes an armature 32 fastened to a support cylinder
216 of the upper piston end member 126'. The alterna-
15 tor armature includes a depending internally threaded
collar 218 which flares outwardly in a funnel-shaped
section 220 and is joined to a cylindrical sleeve 222
which supports the alternator armature 32. The arma-
ture stator 34 is fastened to the inside surface of
20 the alternator housing 200 in radial alignment with
the transverse midplane of the alternator armature.

A top dome 225 is fastened to the alternator
housing 200 by bolts extending through holes in a
bottom radial flange 226 and aligned holes in the
25 top flange 202 on the alternator housing. The dome
225 encloses a control space 227 which is separated
from the bounce space 18 by a partition 228. The bot-
tom face of the dome 225 is slightly concave to pro-
vide a backing for the diaphragm 31 and includes a
30 spider 229 which provides a backing for the diaphragm
31 while permitting working gas which fills the bounce
space 18 to flow freely between the top face of the
diaphragm 31 and the bounce space 18.



-23-

1 In operation, a pressure wave in the engine
working space causes the engine diaphragm 26 to de-
flect upwardly, displacing hydraulic fluid in the hy-
draulic chamber 14 into the hydraulic cylinder 108
5 where it drives the piston-cylinder 126/28 upwardly.
The valve reed 145 is forced shut against the seat
disc 138 and the exhaust valve reed 178 opens to ex-
haust refrigerant compressed in the compression space
161.

10 The top end 126' of the piston 125 simultane-
ously moves upwardly in the hydraulic cylinder 108',
displacing hydraulic fluid into the hydraulic chamber
16 and flexing the bounce diaphragm 31 upwardly toward
the bounce space 18. The gas compressed in the bounce
15 space acts as a spring, storing energy which is re-
turned to the piston-cylinder 126/28 when it is
driven downwardly on the return stroke.

 The fastigium of the compressor cycle coincides
with the minimum enthalpy of the engine cycle and
20 therefore the coupling between engine and compressor
must account for this inherent mismatch. This cou-
pling is accomplished by providing the piston-cylinder
assembly 126/28 with a mass which absorbs energy from
the hydraulic chamber 14 in the form of inertia (mv^2)
25 which is transferred to the gas in the compression
chamber 161 and also via the diaphragm 31 to the gas
in the bounce space 18.

 At the end of the up-stroke, the piston-cylinder
126/28 is momentarily stationary, all its inertial
30 energy having been converted to gas pressure in the
compression chamber 161 and the bounce space 18, and
electrical energy by the alternator. The energy in



1 the bounce space 18 and some of the energy in the
compression space 161 is now returned to the piston-
cylinder 126/28 by the expansion of the compressed
gas. The piston-cylinder 126/28 moves downwardly,
5 compressing gas in the upper compression chamber 161'
and displacing hydraulic fluid in the hydraulic cylinder
108 which flows into the hydraulic chamber 14 and
pushes the diaphragm 26 into the upper portion of the
working gas compression space. The mass of the moving
10 elements, the spring constants of the gas compression
spaces 161 and 161', and the bounce space 18
are selected so that the natural frequency of the
power piston system is near the natural frequency of
the displacer system. The hydraulic fluid pressure
15 and working gas pressure is adjustable, as explained
in detail below, and the gas spring/damping effect of
the compressor is self-regulating, so the systems may
be held in correct relationship to each other.

The apparatus disclosed can be used as a heat
20 pump in which refrigerant having a low boiling temperature,
such as Freon R22 or the like is compressed by the compressor
and the electrical demands of the system such as blowers,
pumps, and solenoids can be supplied by the linear alternator.
The cold refrigerant enters the case 106 at suction pressure at an
25 inlet fitting 224. It fills the interior of the alternator
housing 200, cooling the stator windings 34, and fills the
interior of the case 106. From there it can be drawn into the
compression chamber at each end of the piston-cylinder assembly
30 where it is compressed and expelled at exhaust pressure through
the pipe 182 and the fitting 196 to the external heat exchangers.

1 The control system for starting the engine and
controlling the power output is shown in Fig. 8.
After a temperature differential is established be-
tween the heater head 20 and the cooler 24, it may be
5 necessary to give the displacer an initial movement
to initiate working gas circulation and start the en-
gine. That movement is given in the system by pres-
surizing both hydraulic chambers 14 and 16 to a pres-
sure higher than the mean pressure of the working gas
10 in the engine and in the bounce space 18. This will
cause the diaphragms 26 and 31 to flex outwardly away
from the hydraulic chambers. The hydraulic pressure
can then be released suddenly causing the diaphragms
to bounce inwardly toward the gas compressor thereby
15 causing a pressure wave in the working space in the
Stirling engine 10 which causes an initial movement
of the displacer.

 The hydraulic chambers 14 and 16 are pressurized
by an oil pump 230 which pressurizes oil in an oil
20 supply line 234 to about 20 psi higher than the mean
hydraulic pressure in the hydraulic chambers 14 and
16. The high-pressure oil supply line 234 from the
oil pump 230 is connected to a starter control 232.
The starter control includes a spool valve having a
25 valve body 235 in which is formed a center oil pas-
sage 236 and two end passages 238 and 240. The spool
valve includes a spool valve element 242 biased in
the start position (to the left) by a spring 244 and
is moved to the normal running position (to the right)
30 by a solenoid 246. An axial passage 247 running
through the valve element 242 enables the element to
move in the valve body 235 without pressure cushions
developing at its ends.

1 The operation of the starter control 232 is as
follows: The pump 230 is energized to pressurize hy-
draulic fluid in the line 234. The spring 244 holds
the starter control valve element 242 in the start
5 position (to the left in Fig. 8) wherein fluid com-
munication is established between the line 234 con-
nected to the end passage 238 through the interior
of the valve body to the center passage 236. The cen-
ter passage 236 is connected to an oil line 252 which
10 links both hydraulic chambers 14 and 16, whereby the
chambers may be pressurized. After the hydraulic
chamber pressure has reached the desired magnitude,
the solenoid 246 is energized and moves the element
242 against the spring force to the left (to the po-
15 sition illustrated in Fig. 8) establishing fluid com-
munication between the oil line 252, through the cen-
ter passage 236 and out the end passage 240 to the
distal portion 253 of the oil supply line, connected
to the high pressure section 234 through a restric-
20 tion 255, and thence to an oil sump, as will be ex-
plained below. This permits the oil pressure in the
hydraulic chambers 14 and 16 to drop suddenly causing
the diaphragm 26 to flex away from the engine working
space and causing a sudden drop in the pressure of
25 the working gas. The gas spring 58 will sense that
pressure drop somewhat slower than the displacer top
and bottom faces and therefore the displacer will
move downwardly in the shell 46, displacing working
gas through the regenerator to the cooler, thus start-
30 ing the working gas circulation and the engine cycle.

The oil line 252 also serves to establish com-
munication between the hydraulic chambers 14 and 16
at the midstroke position of the piston cylinder as-
sembly 126/28 to equalize the pressure in the two



1 chambers. As shown in Fig. 1A, a passage 254 leads
from an opening in the well 132 of the piston end
sections 126 through a web (not shown) in the piston
end member 126 and out through an opening 256 in the
5 wall of the piston end member 126. A corresponding
opening 258 in the hydraulic cylinder liner 114 opens
to an annular groove 259 in the inside wall of the
hydraulic cylinder 108 which in turn leads to an oil
passage 260 in a web 262 extending from the hydraulic
10 cylinder 108 and the wall of the case 106. The open-
ing 258 in the cylinder liner 114 is aligned with the
opening 256 in the piston wall at the midstroke posi-
tion of the piston-cylinder assembly, thereby es-
tablishing communication through the oil passage 260,
15 through the midstroke balancing line 252 connected to
a connector 263 on the wall of the case 106, and
through a corresponding passage 264 through the al-
ternator housing 200 to a corresponding midstroke
balancing oil passage system in the top piston end
20 member 126'. This passage system permits the oil
pressure in the two hydraulic chambers 14 and 16 to
equalize at the midstroke position of the piston-
cylinder assembly 126/28 to ensure dynamic centering
of the midstroke position in the housing, and the al-
25 ternator armature 32 in the stator 34.

The mean hydraulic fluid pressure in the hy-
draulic chambers 14 and 16 must be equal to the mean
working gas pressure in the compression space in the
engine and the bounce space 18 in the top end section
30 225. To maintain this equality, a pressure control
270 is provided having a body 272 defining therein a
cylindrical chamber which houses a cylindrical
plunger 276. One end 274 of the plunger 276 is con-
nected to a long roll diaphragm 278 such as a

1 Bellowfram, and the other end 275 controls an oil
drain port, as will be explained presently. The
Bellowfram separates the chamber into two ends: one
5 end 279 is connected by a capillary gas line 280 to
the bounce space 18 to insure that the gas pressure
behind the Bellowfram 278 is at the engine working gas
mean pressure. The other end 281 of the chamber is
connected to the oil line 253 and thence through the
restriction 255 to the high-pressure oil supply line
10 234. The restriction 255 normally reduces the hy-
draulic pressure to about the mean fluid pressure in
the engine.

The end 275 of the plunger 276 is disposed near
the oil drain port 282 in the wall of the chamber 281
15 which can be covered and uncovered by the plunger 276.
When the pressure of the working gas is higher than
the hydraulic pressure, the pressure in the gas end
279 of the pressure control 270 is higher than the
pressure in the fluid end 281 and moves the plunger
20 276 toward the end 281, sealing off the drain port
282. The normally open passage through the start con-
trol 232 between the end passage 240 and the center
passage 236 permits the pump 230 to raise the pressure
of the hydraulic fluid through the restriction 255 in
25 the hydraulic chambers 14 and 16 during the midstroke
position of the piston cylinder assembly until the hy-
draulic and working gas pressures are equal.

When the hydraulic fluid pressure is higher than
the mean working gas pressure, the plunger 276 moves
30 toward the gas end 279 of the pressure control 270,
uncovering the drain port 282 and permitting hydraulic
fluid to bleed out of the fluid end chamber 281. The
back pressure is thus relieved and hydraulic fluid

-29-

1 can flow from the chambers 14 and 16 through the mid-
stroke pressure balancing line 252, the starter con-
2 control valve, and the oil line connecting the end pas-
sage 240 to the line 253 downstream of the restriction
5 255, until the hydraulic fluid pressure and the work-
ing gas pressure are equalized.

Power modulation is achieved by controlling the
pressure of the working gas in the engine. Essen-
tially, the technique for controlling the gas pressure
10 in the working space is to selectively connect the
control volume 227 in the top section 225 of the ves-
sel to the working space through a check valve which
permits gas to flow in the desired direction during
the portion of the cycle in which the space to receive
15 the gas is at a lower pressure than the space from
which the gas is supplied. For example, if it is de-
sired to lower the pressure of the working gas in the
working space of the engine, a power modulation con-
20 trol 290 will permit gas flow from the working space
to the control volume 227 during the high-pressure
periods of the engine cycle, but prevent flow of gas
in the opposite direction from the control volume.

The power modulation control 290 includes a
body 292 having a pressure increase solenoid 294
25 mounted on one end and a pressure decrease solenoid
296 mounted on the other end. The solenoids 294 and
296 are connected to a control element 298 which
slides axially in a bore 300 in the power modulation
control body 292. A pair of centering springs 302
30 and 304 bear against opposite shoulders on the control
element 298 to center the element in the bore when the
two solenoids are deenergized. The center portion of
the control element 298 is relieved at 306 to provide



1 gas flow between a center port 307 and a right port
309 or left port 311 when the control element is dis-
placed to the right or left, respectively, in the bore
300, but prevent gas flow when the element is centered
5 An inflow check valve 308 and an outflow check valve
310 are provided to permit the flow of gas into and
out of the control body 292 depending on the position
of the control element 298. The center gas port 307
from the control element body 292 is connected to the
10 control volume by a fluid line 314. The right and
left ports 309 and 311 are connected by fluid lines
to the working space by a fluid line 316.

In operation, when it is desired to increase
the power of the system, the increase solenoid 296 is
15 energized pulling the control element 298 to the left
in Fig. 8, thereby establishing fluid communication
between the lines 314 and 316 through the check valve
310 and the control body passages to permit fluid
flow from the control chamber through the relieved
20 portion 306 in the control element and hence through
the fluid line 316 into the bounce space 18 at the
working space in the Stirling engine and the engine
working space. The fluid flow occurs only during the
low-pressure portions of the Stirling engine cycle
25 since the control chamber pressure is lower than the
maximum cycle pressure of the gas in the working
space. The engine working gas pressure is thus in-
creased which increases the engine power.

When it is desired to decrease the power in the
30 Stirling engine, the decrease solenoid 294 is ener-
gized to pull the control element 298 to the right
against the force of the spring 304. Communication
is established between the inlet check valve 308 and

1 the central gas passage 307 so that fluid can flow
from the Stirling engine working space during the
high-pressure periods of its cycle through the check
valve 308 and the control passage to the central pas-
5 sage 307 and thence through the line 314 into the
control chamber 227. The pressure of the working gas
in the Stirling engine and the bounce space is thus
reduced, and the engine power is reduced.

The system described above provides a Stirling
10 engine powered compressor-alternator which is sealed
to prevent the loss of working gases and lubricant,
and is provided with positive internal sealed separa-
tion of the engine working fluid from the compressor
working fluid. The sealing is achieved by diaphragms
15 operating in hydraulic chambers to give an incompressible
linkage between the power piston and the diaphragm,
without creating a stress concentration zone on the diaphragms.
The engine cycle and the compressor cycle are made concordant by the mass and damping
20 of the power piston and linear alternator armature.
The system power output and internal pressure balancing
are automatically controlled, making possible
continuous power modulation in response to external
power demand. The internal electrical power require-
25 ments are provided by the linear alternator, making
the system completely independent of the vulnerable
external grid so that a gas fuel source is the only
energy requirement.

Obviously, numerous modifications and variations
30 of the above described preferred embodiment are possible
in view of this disclosure. It is, therefore,
to be expressly understood that these modifications
and variations, and the equivalents thereof, may be

- 1 practiced while remaining within the spirit and scope of the invention which is defined by the following claims, wherein we claim:



Claims :

- 1 1. A Stirling engine power unit, comprising:
a vessel adapted to hold a working gas under high
pressure in a working space;
- 5 an external heater for heating the working gas in a
hot space of said vessel, and a cooler for cooling
the working gas in a cool space of said vessel;
- a displacer movable in said vessel to shuttle the
working gas between said hot space and said cool space
to produce a periodic pressure wave in the working gas;
- 10 a first flexible wall having one face sealing the work-
ing gas in said working space and flexing in response
to said pressure wave;
- a first hydraulic chamber adapted to contain a hydrau-
lic fluid and sealed on one side by the other face of
15 said first flexible wall, so that flexing of said first
flexible wall causes displacement of hydraulic fluid in
said hydraulic chamber;
- a power piston having one end in said first hydraulic
chamber and movable in response to displacement of the
20 hydraulic fluid in said hydraulic chamber;
- a second hydraulic chamber adapted to contain a hydrau-
lic fluid, bounded on one side by the other end of
said power piston, and bounded on the other side by
one face of a second flexible wall;
- 25 a bounce space adapted to be filled with said working
gas, said bounce space bounded on one side by the
other face of said second flexible wall and on the
other side by an interior surface of said vessel.

- 1 2. The engine defined in claim 1, wherein:
- a control plenum is formed in said vessel adapted to
 be filled with said working gas; and
- means for modulating the power of the engine with gas
5 from said control plenum.
3. The engine defined in claim 2, wherein:
- said power modulation control includes a gas flow
 control connected to said working space and said con-
 trol plenum for selectively increasing the working gas
10 pressure in said working space for increasing the
 engine power, and decreasing the working gas pressure
 in said working space for decreasing the engine power.
4. The engine defined in claim 1, further comprising:
- a pressure balance control for maintaining a selected
15 pressure proportion between said hydraulic fluid and
 said working gas.
5. The engine defined in claim 1, further comprising:
- a starter control for producing a starting pressure
 wave in said working gas to move said displacer and
20 thereby initiate working gas circulation.
6. The engine defined in claim 3, wherein said power
 modulation control further comprises a gas flow line
 for connecting said control plenum to said working
 space, and a pair of check valves which selectively
25 permit working gas to flow through said gas flow
 line between said control plenum and said working
 space at high and low portions of said periodic

- 1 pressure wave in said working space.
7. The engine defined in claim 6, wherein said power modulation control further comprises a solenoid actuated spool valve for selectively connecting said
- 5 check valves in said gas flow line to select the direction of gas flow in said gas flow line.
8. The engine defined in claim 5, wherein said starter control further comprises:
- 10 a hydraulic fluid pump for creating a high-pressure source of hydraulic fluid;
- a hydraulic fluid sump for creating a high-pressure source of hydraulic fluid;
- a hydraulic fluid sump at low pressure;
- 15 valve means for connecting said first hydraulic chamber through a hydraulic fluid flow path to one of said source and sump to flex said first flexible wall in one direction, and for suddenly connecting said first hydraulic chamber to the other of said source and sump to quickly flex in the other direction to create said starting pressure
- 20 wave in said working gas.
9. The engine defined in claim 8, wherein:
- said valve means is a spool valve movable axially in a housing to selectively connect said first hydraulic chamber to said source and to said sump, said spool
- 25 valve being arranged to initially connect said first hydraulic chamber to said high-pressure source, and then, in the starting and running configuration, connect said first hydraulic chamber to said sump.



1 10. The engine defined in claim 8, wherein said hydraulic
fluid flow path includes a center port system for
establishing fluid flow when said power piston is at
the center position thereof, and for cutting off said
5 fluid flow at all other positions of said power piston.

11. The engine defined in claim 4, wherein said pressure
balance includes: a housing defining a chamber; a
piston movable in said chamber; a hydraulic space at
one end of said chamber adapted to receive hydraulic
10 fluid under the mean hydraulic pressure in the engine
to move said piston in one direction in said chamber;
a gas space at the other end of said chamber adapted
to receive working gas under the mean working gas
pressure in the engine to move said piston in the
15 other direction in said chamber; and inlet port in
said gas space and a gas line connected between said
inlet port and said working space to pressurize said
working space with working gas at mean engine working
gas pressure; an inlet hydraulic fluid port in said
20 hydraulic fluid space and an outlet port in said
hydraulic fluid space adapted to be covered and
uncovered by said piston when the force exerted on
said piston in said one direction is greater and less
than the force exerted on said piston in the other
25 direction, respectively; inlet and outlet fluid lines
connected, respectively, to said hydraulic chambers
at the midstroke position of said power piston, and
to a hydraulic fluid sump, respectively.

12. A free-piston Stirling engine, comprising:
30 a hermetically sealed vessel enclosing a working space
adapted to contain a working fluid;

a heater for heating said working fluid;



1 a regenerator for removing heat from said working fluid, storing heat, and later returning the heat to the working fluids;

a cooler for cooling said working fluid;

5 a displacer axially movable in said working space for shuttling said fluid between said heater and said cooler through said regenerator to produce a pressure wave in said working space;

10 a flexible power diaphragm extending across and sealing said working space;

15 said power diaphragm having a displacement volume ΔV produced by motion between the fully flexed extremities of the range of travel of said diaphragm greater than about two cubic inches in the pressure of a pressure swing in said engine working fluid greater than about 80 psi to produce a power transfer capacity through said diaphragm in excess of about one Kilowatt at about 60 Hz.

13. The engine defined in claim 12, further comprising:

20 a power cylinder in fluid communication with said power diaphragm;

a power piston in said cylinder;

means for filling said power cylinder between said piston and said diaphragm with hydraulic fluid;

25 whereby pressure waves generated in said vessel by cyclic transfer of said working fluid between said vessel ends are transmitted through said diaphragm



1 and displace said hydraulic fluid to drive said power piston.

14. The engine defined in claim 13, further comprising:

a second piston linked to said power piston;

5 a bounce diaphragm in fluid communication with said second piston;

a second hydraulic chamber defined between said bounce diaphragm and said second piston;

10 means for filling said second space with hydraulic fluid;

a sealed bounce space bounded in part by the other face of said bounce diaphragm and containing a compressible fluid;

15 whereby motion of said power piston in said one direction is transmitted to said second piston, through the hydraulic fluid in said second space and through said bounce diaphragm to compress said compressible fluid and thereby to store energy usable to drive said power piston back in the other direction
20 for the return stroke.

15. The engine defined in claim 14, further comprising a fixed piston fixedly mounted to said vessel and having two valves mounted on the ends of said fixed piston;

25 said power piston including a compressor cylinder connected to said power piston and reciprocating therewith, said compressor cylinder mounted on said



- 1 fixed piston in telescoping relationship and having
two valves mounted on the ends of said compressor
cylinder;
- 5 said compressor cylinder having an end plate mounted
on each end of said compressor cylinder said end
plates each having a gas check valve therein;
- said fixed piston having an end adjacent each end of
said compressor cylinder, each said fixed piston end
having a gas check valve therein;
- 10 two compression chambers, one each defined between the
ends of said fixed piston and the adjacent ends of the
compressor cylinder;
- whereby a double-acting compressor is provided having
two compression strokes per engine cycle.
- 15 16. The engine defined in claim 15, wherein:
- said fixed piston is mounted on a compression gas
outlet pipe having an internal passage communicating
with a connector for connection to external gas lines;
- 20 a suction plenum is defined between said vessel and
said power piston;
- said vessel has connected thereto a gas inlet fitting
for admitting suction pressure gas into said suction
plenum;
- 25 said gas check valves on said compressor cylinder
are inlet valves;
- said gas check valves on said fixed piston ends are



1 exhaust valves;

whereby the gas flow area for the suction gas is greater than the gas flow area of the compression gas.

5 17. The engine defined in claim 16, further comprising heat insulation in said compression gas outlet pipe to retard the conduction of heat from the hot compression gas to the cold suction gas.

18. The engine defined in claim 12, wherein:

10 said power diaphragm lies in a plane perpendicular to the axis of said displacer and about coaxial therewith.

19. A free-piston Stirling engine, comprising:

a hermetic case;

15 a working space within said case;

means for admitting a pressurized working fluid into said working space;

20 means for heating said working fluid at one end of said working space, and means for cooling said working fluid at the other end of said space;

a fluid path for transfer of said working fluid between said ends of said working space;

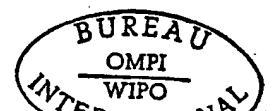
a regenerator in said fluid path;

a displacer in said working space and movable axially



-41-

- 1 therein for shuttling working fluid through said
 fluid path cyclically between said ends of said
 working space to cause cyclic changes of working
 fluid pressure;
- 5 a diaphragm having an inner face extending across
 and sealing said other end of said working cylinder;
- resilient means between said displacer and said case
 for exerting a force on said displacer tending to move
 said displacer toward said one end when said displacer
10 is at said other end;
- a hydraulic chamber in fluid communication with the
 outer face of said diaphragm and adapted to be filled
 with hydraulic fluid;
- a hydraulic cylinder in fluid communication with said
15 hydraulic chamber; and
- a hydraulic piston in said hydraulic cylinder;
- whereby said cyclic changes of pressure in said work-
 ing fluid are transmitted through said diaphragm to
 the hydraulic fluid in said hydraulic chamber and
20 hydraulic cylinder to cause movement of said piston.
20. The free-piston Stirling engine defined in claim 19,
 wherein said hydraulic piston oscillates around a
 stationary piston fixed in position relative to said
 vessel, forming with said stationary piston a gas
25 compressor.
21. The engine defined in claim 20, wherein said station-
 ary piston has attached to the end thereof an exhaust
 valve that communicates with a gas compression chamber



- 1 defined between the end of said stationary piston
and the internal end of said hydraulic piston;
- an exhaust conduit communicating with said exhaust
valve and extending through said stationary piston
5 and through said vessel to an outlet connection on
the exterior of said vessel.
22. The engine defined in claim 21, wherein said station-
ary piston has another end face which acts in a
second gas compression space defined within another
10 portion of said power piston, whereby said power pis-
ton produces two compression strokes for each cycle
of the engine.
23. A Stirling engine driven compressor, comprising:
- a vessel defining therein an engine working space
15 adapted to contain an engine working gas;
- means for heating the working gas contained in one
portion of said working space;
- means for cooling the working gas contained in
another portion of said working space;
- 20 a displacer movable axially in said working space to
displace working gas cyclicly between said heating
and cooling means to create a pressure wave in said
working gas;
- a power piston moved by said pressure wave;
- 25 a gas compressor having a compression space defined
by a variable volume chamber within said vessel;



- 1 intake and exhaust valves for admitting and exhaust-
ing gas to be compressed and compressed gas, respec-
tively, into and out of said variable volume chamber;
- 5 said variable volume chamber having at least one
movable surface driven by said power piston for
changing the volume of said variable volume chamber
and compressing gas in said compression space;
- 10 a mass linked to said one movable surface for stor-
ing energy during high-energy output periods of said
engine cycle and delivering said stored energy to
said compression member during high-power require-
ment periods of the compressor cycle.
24. The free-piston Stirling engine defined in claim 23,
further comprising a linear alternator armature and
15 stator, relatively movable, and comprising a portion
of said mass.
25. The free-piston Stirling engine defined in claim 23,
wherein said variable volume chamber further com-
prises a fixed surface on one end of a stationary
20 piston fixed in position relative to said vessel in
telescoping relation to said power piston.
26. The engine defined in claim 25, wherein said
stationary piston has attached to said one end
thereof an exhaust valve that communicates with
said gas compression space, and an exhaust conduit
25 extends through said stationary piston and through
said vessel to an outlet connection on the exterior
of said vessel.
27. The engine defined in claim 26, wherein said power
piston includes a first end in power transfer rela-
30 tion to said engine working space, and a second end

- 1 in power transfer relation to a bounce space.
28. The engine defined in claim 27, wherein said
stationary piston has another end face which acts
in a second gas compression space defined within
5 another portion of said power piston, whereby said
power piston produces two compression strokes for
each cycle of the engine.
29. A free-piston Stirling engine compressor-alterna-
tor, including:
- 10 a. a hermetically sealed vessel defining therein
a working space including, in operation, a hot
zone and a cold zone and adapted to contain a
working gas;
- 15 b. means for heating the working gas in said hot
zone;
- 20 c. means for cooling the working gas in said cold
zone;
- 20 d. a displacer movable in said working space for
shuttling working gas between said hot and
cold zones through a regenerator for alternately
heating and cooling said gas and generating a
cyclic pressure wave in said gas;
- 25 e. an engine diaphragm sealing said working space
and flexing in response to said pressure waves
in the working gas in said working space;
- 25 f. a first hydraulic chamber bounded on one side by
said engine diaphragm and adapted to contain a
hydraulic fluid;

- 1 g. a piston having a portion movable in a first hydraulic cylinder, said piston having two faces, one of which is hydraulically coupled to said first hydraulic chamber;
- 5 h. a second hydraulic cylinder receiving another portion of said piston and defining a portion of a second hydraulic chamber adapted to contain a hydraulic fluid, bounded on one side by the other face of said piston and on the other side by a second flexible diaphragm;
- 10 i. a gas compression chamber having a moving face driven by hydraulic fluid pressure in one of said hydraulic chambers and a second face opposed to said moving face for compressing gas therebetween;
- 15 j. valve means in said gas compression chamber for admitting gas to be compressed, and exhausting compressed gas;
- 20 k. a mass coupled to said hydraulic cylinders for storing energy from said engine working gas pressure wave and delivering said energy to said compression chamber moving face.
- 25 30. The structure defined in claim 29, further comprising a sealed bounce space adapted to contain a gas bounded in part by said second diaphragm, whereby kinetic energy of said piston is stored as compression energy in the gas in said bounce space when said piston is driven toward said bounce space by said pressure wave, and said compression energy is
- 30 returned to said piston on the return stroke of said

1 piston for the compression of said working gas on the compression stroke of the Stirling cycle.

31. A free-piston Stirling engine, comprising:
a vessel having a closed end and an open end, and
5 defining therein a working space adapted to be filled with a working gas;

means for heating said working gas at said closed end, and means for cooling said working gas at said open end;

10 a displacer disposed within said working space and reciprocally movable axially therein to displace working gas between said open end and said closed end to cause a cyclic pressure wave to occur in said working fluid;

15 means for:

- a. mounting and supporting said displacer for axial movement in said working space,
- b. storing energy upon movement of said displacer toward said open end of said vessel,
- 20 c. returning said energy to said displacer by driving it toward said closed end of said vessel,
- d. reducing the effective area of said displacer end, adjacent said open end of said vessel on which said working gas can act to create a pressure force imbalance which is exerted on said
25 displacer and varies cyclically with the pressure wave in said working gas, serving with said energy storage and return means to compensate for windage and friction losses of said displacer

- 1 and maintain the axial oscillation thereof;
- a flexible engine diaphragm having an inside face and an outside face, said inside face sealing said open end of said vessel;
- 5 said diaphragm flexing outward in response to increases in pressure of said working gas in the course of said cyclic pressure wave;
- a hydraulic chamber adapted to be filled with hydraulic fluid and bounded on one side by said outside face of said engine diaphragm;
- 10
- a power piston having a face bounding the other side of said hydraulic chamber and movable in a hydraulic cylinder, said piston being driven by hydraulic fluid displaced in one direction from said hydraulic chamber by said engine diaphragm;
- 15
- an energy storage device for storing energy of said piston upon movement thereof driven by said hydraulic fluid, and for returning a portion of said energy back to said piston to displace said hydraulic fluid, flex said engine diaphragm inwardly, and compress said working gas.
- 20
32. The engine defined in claim 31, wherein said displacer energy storage and return means includes:
- a resilient medium operatively bearing against said vessel and said displacer, whereby said medium, when compressed, exerts a force on said displacer and exerts the reaction force on said vessel.
- 25

1 33. The engine defined in claim 32, wherein:

5 said mounting, supporting, storing, returning and
reducing means includes a base member attached to
said vessel at said closed end, a post attached to
said base member and projecting therefrom toward
said hot end, and a well formed in the end of said
displacer adjacent said open end of said vessel,
said well having a diameter to receive said post
with a sliding fit, said well having a closed end
10 which with said port forms a gas spring between said
displacer and said vessel;

15 said post, where it enters the end of said displacer,
reducing the effective area of said displacer end on
which said working gas can act thereby creating said
pressure force imbalance.

20 34. A free-piston Stirling engine having a sealed vessel
defining therein a working space; a displacer
axially movable in the working space for shuttling
working gas between one end of the working space
where it is heated, and the other end of the working
space where it is cooled; a power piston axially
movable in said vessel having a power stroke under
the influence of the expansion of said working gas
at a high temperature, and a compression stroke in
25 which it compresses said working gas at a low temper-
ature; wherein improvements comprise:

means for hermetically separating one end of said
power piston from said working space;

said power piston having a second end;

- 1 a bounce space connected to said working space by connection means that allows flow of working gas only fast enough to equalize the mean gas pressure;
- 5 said power piston second end being in power transfer relation to said bounce space;
- means for hermetically separating said second end of said power piston from, and for sealing said bounce space.
- 10 35. The engine defined in claim 34, wherein said separating means includes a diaphragm extending across and sealing the portion of said vessel containing said working space.
- 15 36. The engine defined in claim 35, wherein said separating means further includes a hydraulic chamber bounded on one side by said diaphragm and on the other side by one face of said power piston.
37. A free-piston Stirling engine/compressor, comprising:
- 20 a hermetic vessel enclosing an engine working space adapted to contain a working gas;
- a heater for heating the working gas and a cooler for cooling the working gas; a regenerator for extracting and storing heat from the working gas as it flows in one direction, and returning the heat to the working gas as it flows in the other direction;
- 25 a displacer reciprocally mounted in said working space to displace a working gas between said heater and said cooler through said regenerator to create a pressure wave in said working gas;

1 an engine diaphragm extending across and sealing said
working space and having one face in communication
with said working gas;

5 a hydraulic chamber bounded by and sealed by the
other face of said engine diaphragm;

a power piston reciprocally mounted in said vessel
and having one face defining a movable wall of said
hydraulic chamber;

a second hydraulic chamber;

10 a second face on said power piston defining a second
movable wall of said second hydraulic chamber;

a second diaphragm extending across and having one
face sealing said second hydraulic chamber;

15 a gas space bounded on one side by the other face
of said diaphragm;

control means for adjusting the mean pressure on at
least one side of at least one diaphragm to maintain
a predetermined proportional relationship between the
mean pressure in the volumes on both sides of said
20 diaphragms.

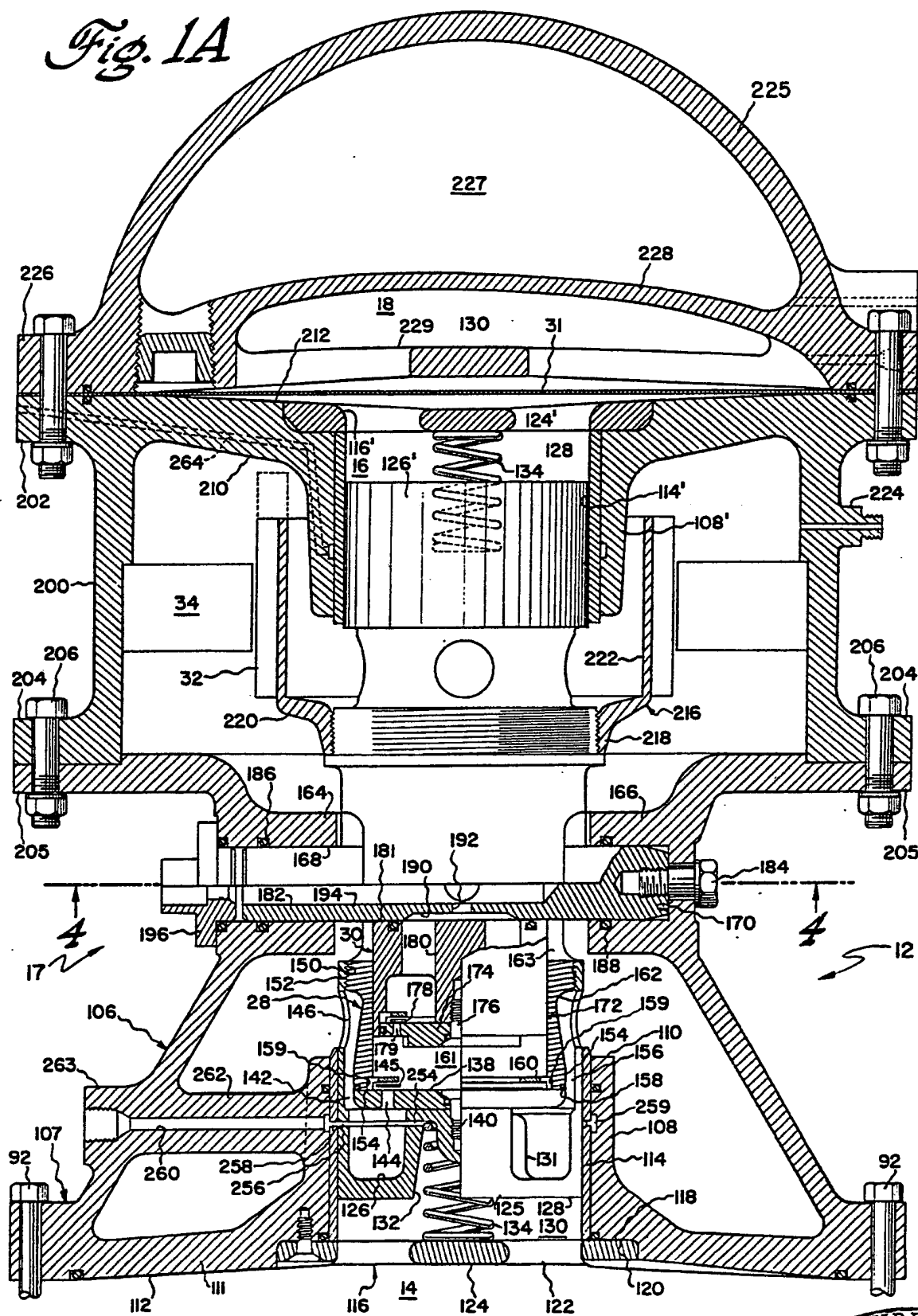
38. The apparatus defined in claim 37, wherein said pres-
sure proportion maintaining means comprises:

25 a pressure comparator having two compartments for
comparing the mean pressures in at least two of said
volumes and movable from a neutral position when the
pressure differential deviates from a predetermined
value;



- 1 a gas duct connecting said engine working space to one of said comparator compartments;
- a second duct for connecting one other of said volumes to the other of said compartments;
- 5 a valve controlled by said pressure comparator;
- a fluid storage volume adapted to contain fluid at a pressure different from said engine working gas mean pressure;
- 10 a fluid conduit connecting said fluid storage volume and one of said volumes when said pressure comparator moves from said neutral position and moves said valve to allow fluid to flow between said one volume and said fluid storage volume to restore said predetermined proportional relationship.
- 15

1 / 7

Fig. 1A

2 / 7

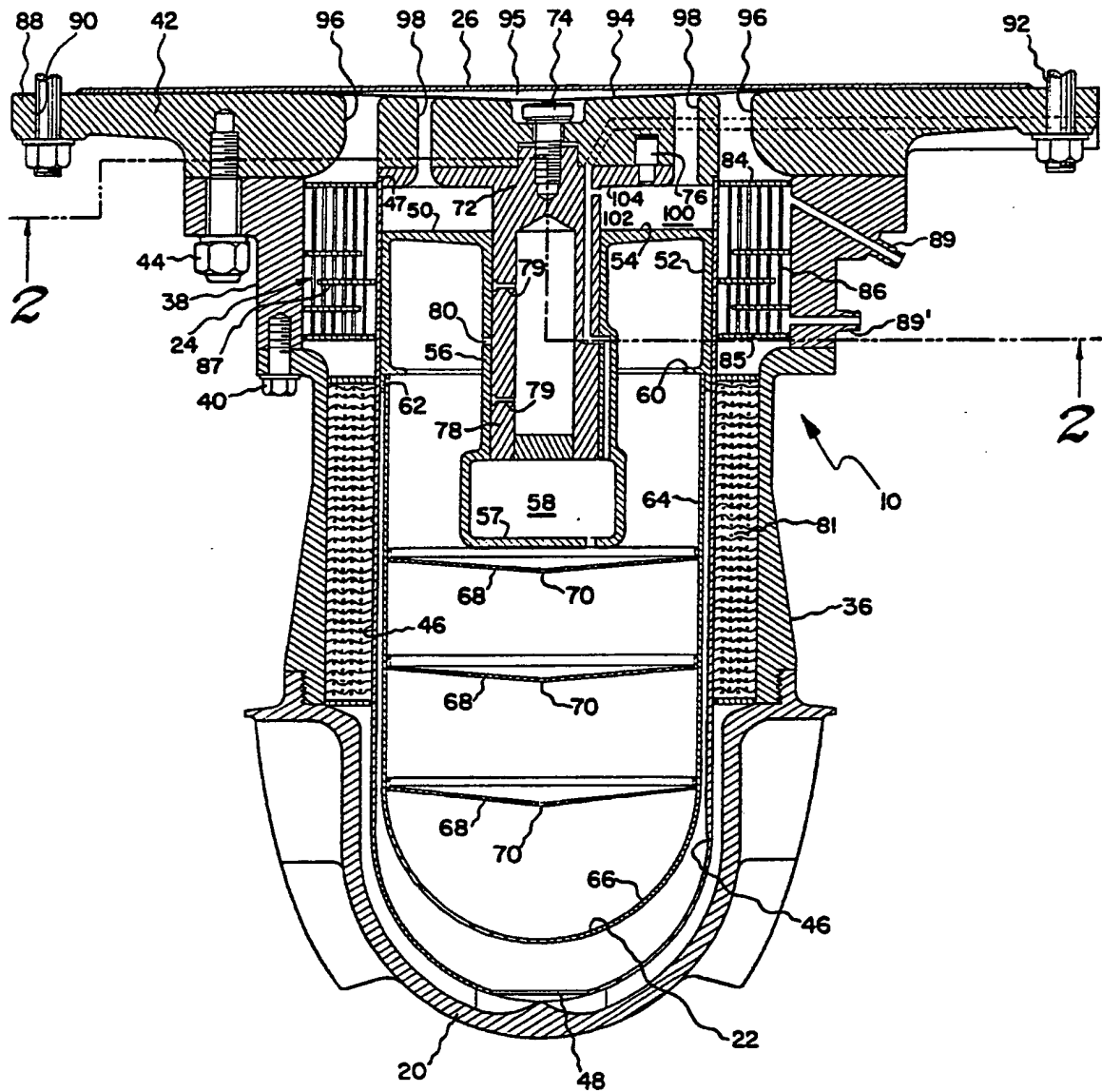


Fig. 1B

4 / 7

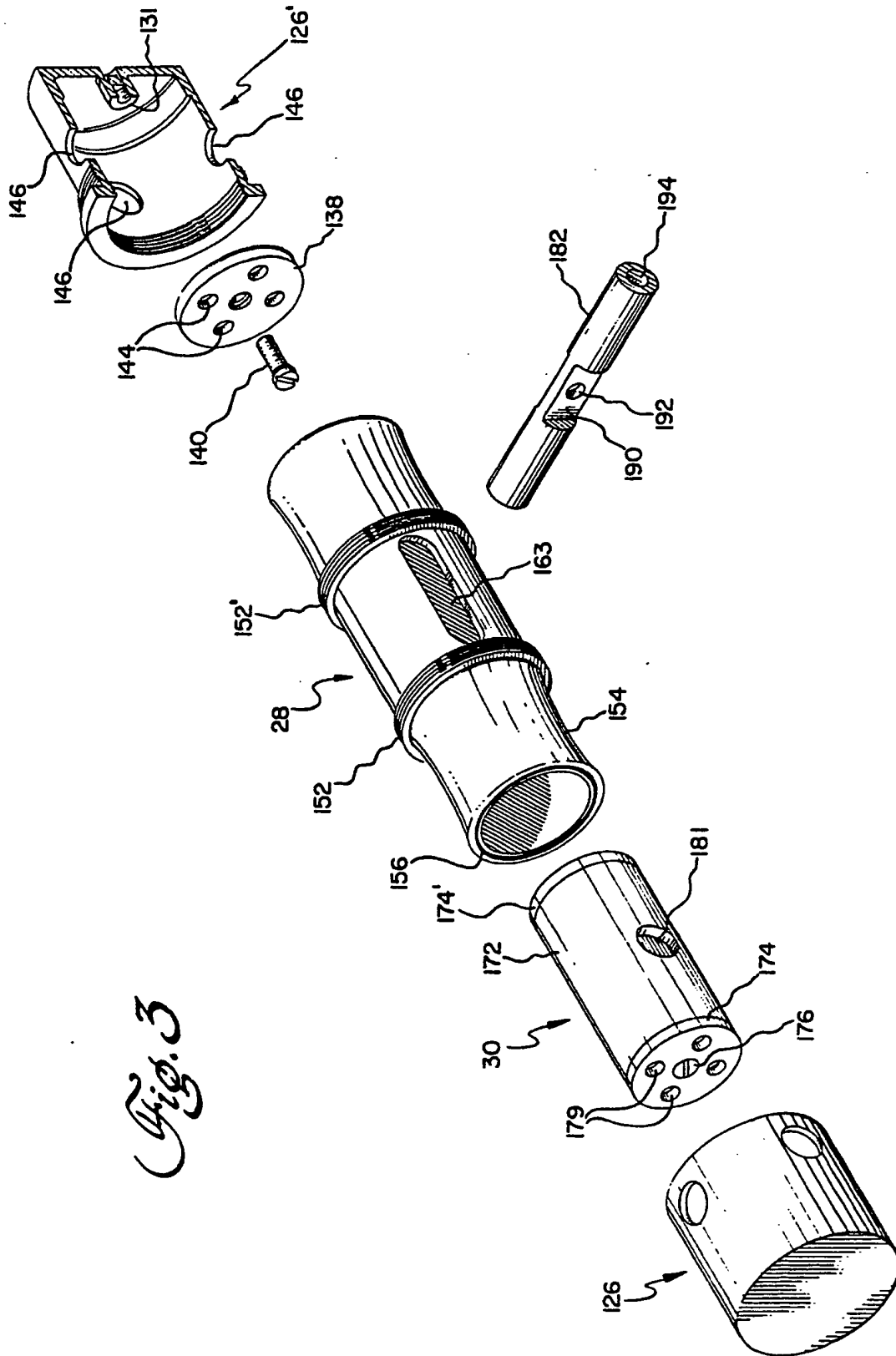


Fig. 3

6 / 7

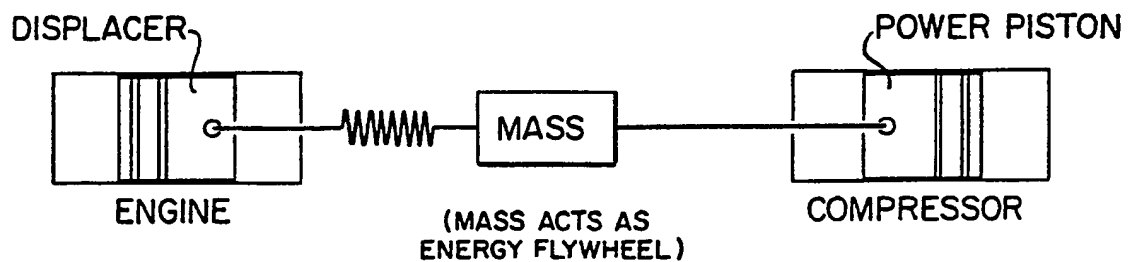
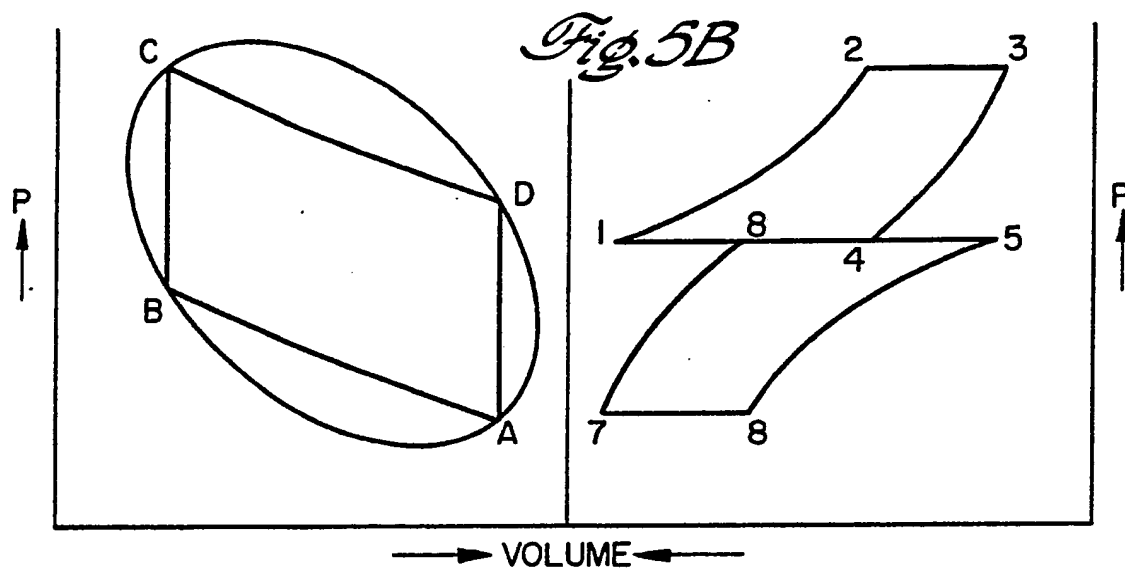
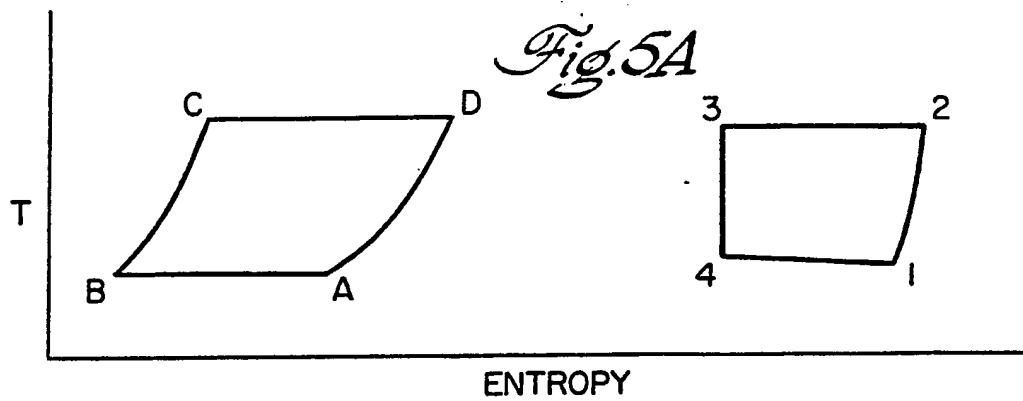


Fig. 5C

7 / 7

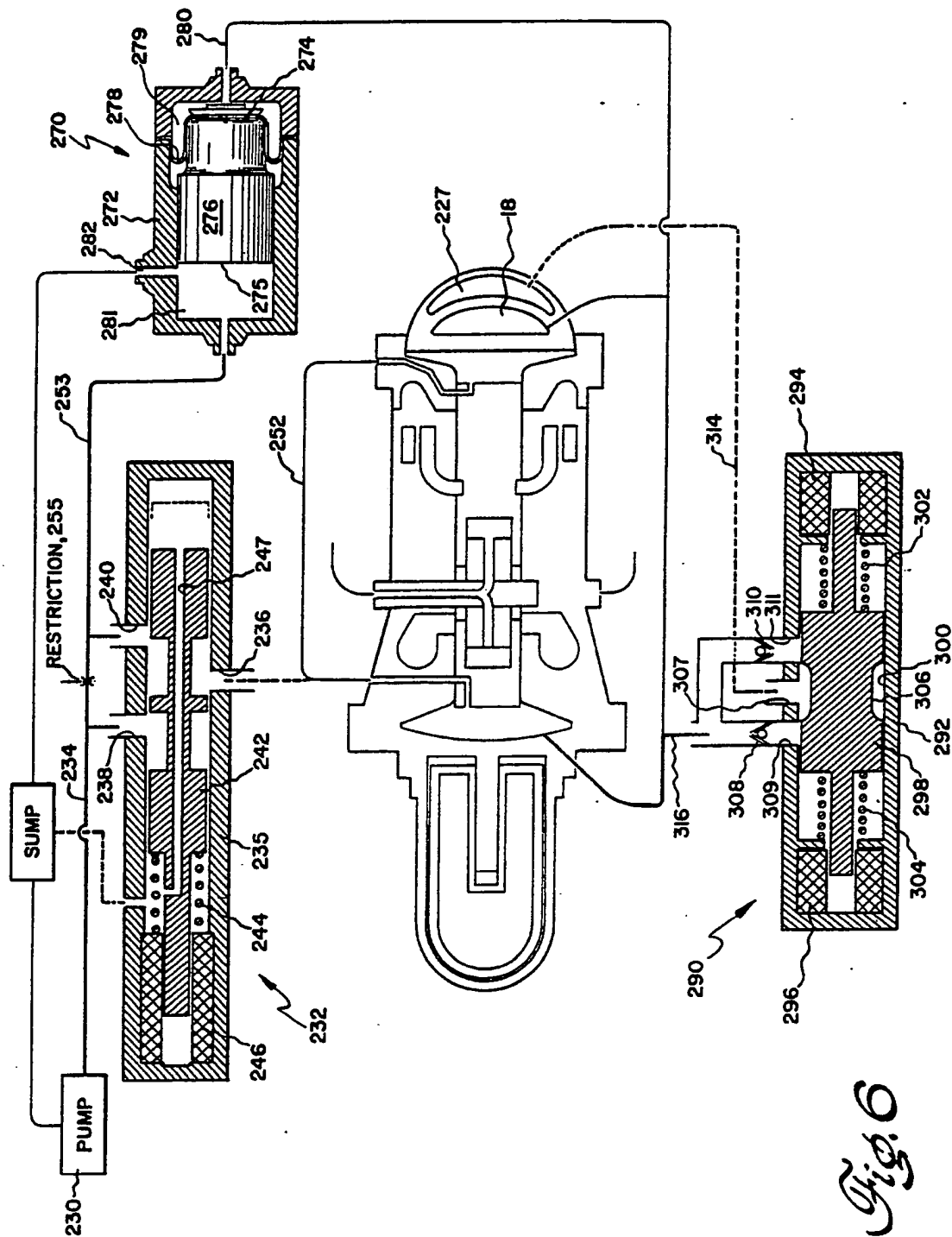


Fig. 6

INTERNATIONAL SEARCH REPORT

International Application No PCT/US 81/00935

I. CLASSIFICATION OF SUBJECT MATTER (If several classification symbols apply, indicate all) *		
According to International Patent Classification (IPC) or to both National Classification and IPC		
INT. CL. F02G 1/04 U.S. CL. 60/520, 526; 62/6		
II. FIELDS SEARCHED		
Minimum Documentation Searched *		
Classification System	Classification Symbols	
U.S.	60/517, 518, 520, 521, 526 62/6	
Documentation Searched other than Minimum Documentation to the Extent that such Documents are Included in the Fields Searched *		
III. DOCUMENTS CONSIDERED TO BE RELEVANT ¹⁴		
Category *	Citation of Document, ¹⁵ with Indication, where appropriate, of the relevant passages ¹⁷	Relevant to Claim No. ¹⁸
A	US, A, 3,822,388 Published 02 July 1974 Martini et al	1-38
A	US, A, 3,828,558 Published 13 Aug. 1974 Beale	1-38
A,P	US, A, 4,215,548 Published 05 Aug. 1980 Beremand	1-38
<p>* Special categories of cited documents: ¹⁶</p> <p>"A" document defining the general state of the art</p> <p>"E" earlier document but published on or after the international filing date</p> <p>"L" document cited for special reason other than those referred to in the other categories</p> <p>"O" document referring to an oral disclosure, use, exhibition or other means</p> <p>"P" document published prior to the international filing date but on or after the priority date claimed</p> <p>"T" later document published on or after the international filing date or priority date and not in conflict with the application, but cited to understand the principle or theory underlying the invention</p> <p>"X" document of particular relevance</p>		
IV. CERTIFICATION		
Date of the Actual Completion of the International Search *	Date of Mailing of this International Search Report *	
05 October 1981	15 OCT 1981	
International Searching Authority *	Signature of Authorized Officer ²⁰	
ISA/US	S. F. Husar S. F. HUSAR	

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